Thermo-Fluid Dynamics Analysis of a Building Rock Thermal Storage System

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Thermo-Fluid Dynamics Analysis of a Building Rock Thermal Storage System

Final Report for Mechanical Project 478
Department of Mechanical and Mechatronic Engineering
University of Stellenbosch

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2010
## Executive Summary

### Title of Project

The thermo-fluid dynamics analysis of a building rock thermal storage system.

### Objectives

The thermal and fluid dynamics analysis of the US Sustainability Institute’s School of Public Management and Planning under floor thermal rock storage facility is the primary objective of this project. Computational fluid dynamics analysis of the system is also included in the project. Recommendations regarding the operation of the rock storage facility at Lynedoch are also given.

### Which aspects of the project are new/unique?

The computational fluid dynamics analysis of a thermal rock storage facility.

### What are the findings?

The acceptable correlation between the analytical and measured results both for the test model and the rock store at Lynedoch.

### What value do the results have?

The results will assist in the design of future thermal rock storage facilities. The results will also allow the facility to gain the maximum benefit from its thermal rock storage facility. The results can assist in promoting understanding of the functioning of such a system.

### If more than one student is involved, what is my contribution?

Not Applicable.

### Which aspects of the project will carry on after completion?

None is envisaged.

### What are the expected advantages of continuation?

Not Applicable.

### What arrangements have been made to expedite continuation?

Not Applicable.
Department of Mechanical and Mechatronic Engineering
Stellenbosch University

Declaration

I know that plagiarism is wrong.

Plagiarism is to use another’s work (even if it is summarised, translated or rephrased) and pretend that it is one’s own.

This assignment is my own work.

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Student no: ............................... 

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Date: .................................
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<td>Temperature difference</td>
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<td>$\Delta t$</td>
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<td>$\Delta x$</td>
<td>Length of segment of bed, m</td>
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<td>$\varepsilon$</td>
<td>Void fraction (porosity) of a material</td>
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<td>Dynamic fluid viscosity, kg/ms</td>
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<td>$\nu$</td>
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<tr>
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<tr>
<td>$V$</td>
<td>Velocity, m/s</td>
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$V_f$  Void Volume, $m^3$

$V_o$  Total Volume, $m^3$
1. Introduction

1.1 Motivation

Thermal rock storage facilities, when combined with ventilation systems, have the potential to offer a low energy usage ventilation system with some air conditioning ability. If the system can be modelled correctly, its use can be optimised, and the client will receive maximum benefits from the system.

Rock stores can be used for both heating and cooling purposes. During the cooling cycle, cold air is pumped via fans through the large volume of rocks situated under the building floor at night. This is referred to as charging the rock store, and reduces the temperature of the rocks. During the day, as cooling is required, the recovery phase is initialised. Air is pumped through the rock store into the building. Heat transfer takes place between the hot air and the colder rocks, thus cooling the air as the rocks heat up. This is the recovery phase for the cooling cycle. If heating is required, the rock store is charged by feeding hot air from a catchment area located in the roof of the building through the rock store. Thermal energy is transferred from the air to the rocks. The energy stored in the rocks is then used to heat the air supplied to the building once the ambient temperature cools. The large volume of the rocks, and underground isolation, combined with the thermal capacity of the rocks make this method an alternative to conventional air-conditioning systems.

The US Sustainability Institute situated at Lynedoch has a rock store installed below five classrooms of the primary school. The Institute has many innovative systems in place to make it as resource efficient as possible. The successful completion of this project will allow the US Sustainability Institute to gain the maximum benefit from the rock store. The results from the detailed assessment will allow the institute to educate...
employees and the public on the workings and performance levels of such a system.

1.2 Objectives

To numerically model a thermal rock storage facility, the objectives must first be identified along with the tasks involved with reaching those objectives.

The primary objective of the project is to set up an analytical model of the rock storage facility at the University of Stellenbosch Sustainability Institute situated at Lynedoch. Before this can be accomplished, a scaled down model of a rock store needs to be built, tested and analysed. The simplified model will allow for the testing of the measurement equipment and the thermo-fluid dynamic analysis under more controlled conditions. Data acquisition of the facility forms an integral part of successful numerical modelling of the rock store. The knowledge obtained from the test model, in both the charging and recovery cycles, will then be applied to the physical parameters of the rock store. Acceptable correlation between the analytical and test data will complete the primary objective of the project.

The successful analytical modelling of the rock store will allow detailed instructions of the periodic cycling of both the charging and recovery cycles of the rock storage facility to the University of Stellenbosch Sustainability Institute so that they can achieve optimal performance from the rock storage facility.

1.3 Scope

The aim of this project is to set up an analytical model of the thermal rock storage facility at the US Sustainability Institute. This project will focus on quantifying the performance of the system, both thermally and from a fluid dynamics point of view. This project will identify the cost and timescale procedures involved to complete this analysis on schedule.
The design, building and testing of a scaled model will be included in the project. An analytical model will be set up and the analytical results will be compared to the results obtained from the testing of the model. Testing of the under floor rock thermal storage system will then be conducted at the US Sustainability Institute, Lynedoch. Analytical predictions will then be made for the rock store at Lynedoch and the comparisons of results will be discussed.

A numerical model will be created using computational fluid dynamics. The results from this simulation will be compared to the analytical predictions and the comparison will be discussed.

Recommended operating procedures will be included relating to both summer and winter configurations of the rock store. These recommendations will be split into four phases of increasing complexity.

1.4 Literature study
This section is a simplified summary of some of the work that has been done regarding the thermal relationships present in rock storage systems. Theories that have been developed with regards to heat transfer in packed beds are mentioned briefly in order to clarify the origins of the formulas used to analyze the thermal rock storage facility for this project. This is followed by a brief description of the previous work done regarding the pressure drop across a packed bed and the different materials used for thermal storage. A summary of previous conclusions drawn regarding the effects of wall channelling concludes this section.

1.4.1 Previous work regarding the thermodynamics of packed beds
Çengel (2003) expresses the Nusselt number as a simple power-law relation.

\[
\text{Nu} = C \text{Re}^m \text{Pr}^n
\]  

(1.1)

Where C, m and n are constants which depend on the geometry.
Different shaped particles in a packed bed were examined by Singh et al. (2006) and the effects on the Nusselt number and friction factors in a cylindrical packed bed were analyzed. They related the change in Nusselt number to the shape of the particles by defining a sphericity factor for the particles, where a value of unity represents a spherical particle.

Balakrishnan and Pei (1979a) reviewed all work done regarding packed beds before 1979. They studied the radiation heat transfer between particles in the bed as well as conduction between the particles and conclude in a following publication (1979b) that radiation effects between particles only become significant, relative to the conduction between particles, at temperatures above 400K. When conducting higher temperature analysis they compiled a model that took into account the axial thermal conduction, radiation heat transfer, convection heat transfer between the particles and the fluid as well as the effect that convection had on the conduction through the bed.

Heat losses through the walls of packed beds have had a significant influence on test results of packed beds in previous studies. Beasley and Clark (1984) noted that thermal losses through the walls of a packed bed were substantial. They measured the temperatures along the radius of a cylindrical packed bed and commented that high wall thermal conductivity and the large heat capacity of the wall may have contributed to the obtained results.

The effects of humidity in the air have been analysed previously. Chandra and Willits (1981) tested heat transfer coefficients in packed beds and found that for their tests, between 15 °C and 90 °C, the coefficients were not affected significantly by variations in the moisture content of the air.

Schumann (1929) created an analytical model for the heat transfer through packed beds. Experimental results of packed beds have also been presented by Adebiyi et al. (1998) and Schröder et al. (2006). Many
authors like Gunn (1978) and Singh et al. (2006) have presented methods for predicting the thermal characteristics of packed beds based on experimental data. Martin (2005) developed the Generalised Lévêque Equation (GLE) in order to predict the heat transfer coefficient through a packed bed as a function of the pressure drop across the bed. An energy based analysis presented by Dincer et al. (1997) shows the limitations of packed beds for thermal storage as he includes pumping losses and exergy analysis for both charging and recovery cycles.

1.4.2 Previous work regarding the fluid dynamics of packed beds
Ergan (1952) developed what is commonly known as the classical equation regarding the pressure drop across a packed bed. Du Plessis and Woudberg (2008) give a simplified version of the Ergan equation for laminar Newtonian flow in the Darcy regime. Singh et al. (2006) suggested a method for predicting the pressure drop across packed beds based on experimental results. Diedericks (1999) gives details on the representative unit cell (RUC) model which is based on flow between parallel plates.

1.4.3 Previous work regarding materials used for thermal storage
Özkahraman et al. (2004) published results for the density and thermal conductivity of Sandstone (Berea) as 2150 kg/m³ and 2.9 W/mK respectively. Extensive research has also been done on the optimal rock to bed size ratios and types of rocks to consider for various applications but due to the nature of this project these have little relevance.

Edge effects or wall channelling has a negative effect on the accuracy of the measured data for small scale models of packed beds. The fluid velocity near the walls of a packed bed can be up to twice as fast as the average velocity through the bed as shown by Kaviany (1995). The general method to overcome wall channelling in test models is to make use of compressible insulating layers on the walls so that the particles in
the packed bed may be forced into the layer, thereby reducing the effects of wall channelling.
2 Acquisition of measured data for the model

2.1 Description of the test model
Due to the size and location of the rock store at the US Sustainability Institute it was decided that a scaled testing model in a controlled environment would allow more consistent measurements. The processes of the design, manufacture and testing of the scaled testing model is described in this section. This section is concluded by a discussion of the results obtained during the testing of the model.

2.2 Concept generation
Several rough concepts were generated and investigated based on the information acquired regarding the project requirements. The concepts were shortlisted and the most applicable option was refined. The final concept resembles a 1m wide section of the rock store at the US Sustainability Institute.

2.3 Material selection
The models inlet and outlet structures, as well as the casing, are constructed from 12 mm commercial plywood. The casing is lined with 50 mm polystyrene foam to insulate the model from environmental influences. The compressibility of the foam allows for the rocks to be forced into the walls of the test section to reduce the edge effects as much as possible.

The rocks used are sedimentary sandstone (Berea). This type of rock has a density of 2150 kg/m$^3$ and a thermal conductivity of 2,9 W/mK at 27 °C according to Özkahraman et al (2004). A random sample of 100 rocks, as shown in Figure 1, was measured to find the average volume of the rocks in the test section. This was achieved by submerging each of the sample rocks in water and measuring the change in the water level. The average
volume of the rocks used in the model was found to be $911 \times 10^{-6} \text{ m}^3$. This relates to an average mass of 1.96 kg per rock. Due to the rounded nature of the rocks, spheres with a diameter of 120 mm will be used in the mathematical analysis of the model.

![Figure 1: Photograph of the sample rocks](image)

### 2.4 Details of the design of the model

The final concept was drawn up using Autodesk Inventor 2009. Figure 2 shows an isometric view of the fully assembled test model. The rest of this
section explains the details of the design.

![Isometric diagram of testing model](image)

**Figure 2: Isometric diagram of testing model**

The inlet duct of the model is angled at 7° to distribute the air across the test section as uniformly as possible. Inlet air has to pass over an aluminium heat exchanger. Hot water, circulated through the heat exchanger, allows control of the inlet air temperature.

A constant speed fan is situated at the exhaust of the outlet duct of the model. This is to remove any heating effect the fan may have on the test. This also allows a bell mouth to be connected to the inlet duct. The flow rate of the air through the test model is monitored by the bell mouth at the inlet and controlled by a sliding gate mechanism positioned just before the fan.

Four sets of T-type thermocouples monitored the temperatures in the test model. These are located 0.45 m from either end and are positioned 0.7 m apart. Each set consists of four thermocouples positioned 0.3 m from either side and 160 mm from both the top and bottom walls of the model. Thermocouples in the inlet and outlet ducts monitor the inlet and outlet air temperatures. A wet bulb thermometer is used to monitor the humidity of
the inlet air. The positioning of the testing apparatus can be found in Appendix C. The thermocouples are connected to an Agilent 34970A data logger through a 20 port card.

Detailed CAD drawings of the model can be found in the project file. Cutting drawings were given to the plywood supplier to reduce the construction time of the model. These drawings can also be found in the project file.

2.5 Model construction

The model was assembled in the University’s Heat Transfer Laboratory. The following figures summarise the model construction phase of the project.

Figure 3 shows the assembled plywood casing with polystyrene insulation fitted. Thermocouples were installed at the required depths while filling the model with rocks.

![Figure 3: Initial phase of model construction](image)

Rocks were pressed into the side and bottom insulation to reduce edge effects in the model. A sheet of fibrous insulation shown in Figure 4 was used for the same reason on the top of the model.
2.6 Measured results

Four separate measurements were taken for both the charging and discharging cycles and the data was captured using Agilent BenchLink Data logger 3. The temperature distribution in the model required an average of eight hours to reach steady state. Discharging the model required less time as the convection losses from the packed bed accelerated the time required to reach steady state of the discharging cycle.

The Agilent 34970A data logger was set up to acquire readings from all 20 thermocouples at 30 second intervals. The thermocouples were arranged in groups of 4 along the length of the model. The measured values were then averaged between the sets to simplify the analysis of the temperature distribution curves on a graph.

A Betz 5000 Micromanometer was coupled to a bell mouth, with a throat diameter of 50 mm at the inlet of the test model. A variable shutter was built and fixed on the exhaust duct before the fan. The combination of these allowed full control over the flow rate of the air. A mass flow rate of
0.06 kg/s was selected for the testing as this would achieve the same velocity through the rocks as the fans provide at Lynedoch.

The charging of the test model rock store showed a noticeable temperature wave moving through the model. In Figure 5 the input and output temperature curves are shown as well as the averaged results of the four groups of thermocouples. The numbering of the thermocouple sets on the graph start with 1, nearest the inlet, through to 4, nearest the exhaust.

The fan and heat supply were then shut down and the thermal behaviour of the bed was monitored. Figure 6 shows the thermal behaviour of the packed bed with no airflow. Sections 1 and 4 show greater thermal losses than sections 2 and 3 and this can be attributed to the lack of insulation on the inlet and exhaust ducts of the test model.

Figure 5: Temperature distribution during the charging cycle of the model
After the bed had been charged, the fan was turned off for half an hour to allow the air and rock temperatures to reach equilibrium. The heat supply was also turned off before the fan was turned on again. This implied the start of the recovery cycle, the results of which are shown in Figure 7.

Figure 7: Temperature distribution during the discharging cycle of model
Figure 7 shows the behaviour of the bed during the recovery cycle. The lack of insulation on the inlet and exhaust faces of the model lead to thermal losses in sections 1 and 4. These losses can be seen on the graph by the low initial temperature of section 4. It can also be noted that during the first half hour of the discharging cycle, section 4 experiences an increase in temperature as it is charging from the energy stored in sections 2 and 3.
3 Analytical model of a thermal rock store

This section of the report provides details of the equations used to set up the analytical model of a thermal rock store. The equations are followed by results of the analytical simulation of the test model.

3.1 Pressure drop calculations

Singh et al. (2006) presents a method for determining the pressure drop across a packed bed which incorporates a sphericity factor ($\psi$). This sphericity factor is defined as the ratio of the surface area of a sphere, with volume equal to that of the particle, relative to the actual surface area of the particle.

$$\psi = \frac{A_{\text{sphere}}}{A_{\text{actual particle}}}$$  (3.1)

Due to the rounded and irregular nature of the rocks a sphericity factor of 0.9 is assumed.

The relatively low temperatures found in the specific packed bed of this study led to the assumption of a homogeneous (non-varying porosity) packed bed as is assumed and defined by Bear and Bachmat (1991). Terblanche (2006) defines the possible structures for porous media. The rocks in the packed beds of this project are thus classified as granular. Terblanche also defines the porosity ($\varepsilon$) of a porous medium as the ratio of void volume to the total volume of the packed bed as shown in equation 3.2.

$$\varepsilon = \frac{V_t}{V_o}$$  (3.2)
Kaviany (1995) states that the porosity of crushed rock is between 0.44 and 0.45. A porosity of 0.4 is assumed for this project due to the rounded nature of the rocks.

The dimensionless mass flux, as defined by Krane (1987), was calculated as shown in equation 3.3 below.

\[ \frac{G}{A_{cs}} \sqrt{\frac{\rho_f}{ho_a}} = \frac{m_f}{A_{cs} \rho_a} \]  

(3.3)

The mass flow rate \( m_f \) through the test bed was 0.06 kg/s. An environmental temperature \( T_{env} \) of 18 °C and pressure \( p_a \) of 100.5 kPa were used. The cross sectional area \( A_{cs} \) of the model test bed was 0.5 m\(^2\). The gas constant \( R \) for the air is 287 J/kgK.

The particle Reynolds number \( \text{Re}_p \) was calculated as shown in equation 3.4. The dynamic fluid viscosity \( \mu \) of 1.81x10\(^{-5}\) kg/m.s was used to determine the particle Reynolds number.

\[ \text{Re}_p = \frac{GD}{\mu_f(1 - \varepsilon)} \]  

(3.4)

The hydraulic particle diameter \( D \) in equation 3.4 was experimentally determined by to be 120 mm in section 2.2.2. The values defined were then used in a correlation presented by Singh et al. (2006).

\[ f_s = 4.466 \times 10^{-4} \text{Re}_p^{-0.246} \times 0.592 + 0.228 \times 10^{-1} \text{Re}_p^{0.382} \]  

(3.5)

This correlation is then used to determine the pressure drop across the bed as shown in equation 3.6.

\[ \Delta P = \frac{f_s L \rho_g \beta}{\rho_f D} \]  

(3.6)
The ambient air density ($\rho_f$) is 1.2 kg/m$^3$, and the total length of the bed (L) is 3 m. This resulted in a pressure drop across the model test bed of 4.98 Pa for the conditions specified.
3.2 Thermal calculations

Allen (2010) makes use of the “Effectiveness-NTU” method presented by Hughes (1975) which is a numerical approximation of the 1929 Schumann model which was developed to model the thermal behavior of packed beds.

The volumetric heat transfer coefficient \( h_v \) presented by Chandra and Willits (1981) is based on the results that they analysed from testing rocks.

\[
h_v = \frac{1.45 \ Re_p^{0.7} R_f}{\Delta x}
\]  
(3.7)

This equation requires the particle Reynolds Number \( Re_p \) and the particle hydraulic diameter \( D \) as well as the fluid thermal conductivity \( k_f \) which is 0.0288 W/mK for air.

The number of transfer units (NTU) for the packed bed, defined by Hughes et al. (1976), is a definition that applies to the entire packed bed.

\[
NTU = \frac{h_v \Delta x L}{\dot{m}_f c_{pf}}
\]  
(3.8)

A specific heat capacity of the fluid at constant pressure \( c_{pi} \) is 1006 kJ/kg as presented by Çengel (2003).

The effectiveness-NTU equation for an evaporator or a condenser presented by Allen (2010) as found in Mills (1999) was used to determine the change in the fluid temperature through the packed bed. The bed was divided into sections of 300 mm \( (\Delta x) \) along the direction of the flow and allows the temperature for each section to be calculated at the next time step based on the solid and fluid temperatures at the current time step.

\[
T_{F,i+1} = T_{F,i} - \left( T_{F,i} - T_{x,i} \right) \left( 1 - e^{-NTU(\Delta x)/L} \right)
\]  
(3.9)
Duffie and Beckmann (1991) present a time constant (τ) which is used to simplify the equation regarding the thermal behavior of the solid mass in the packed bed as shown in equation 3.10.

$$\tau = \frac{\rho_p c_p (1 - \varepsilon) A_{es} L}{\kappa_p c_p \varepsilon}$$

(3.10)

Allen (2010) presents an explicit equation to calculate the temperature of the solid mass in the packed bed at the next time step based on the Hughes effectiveness-NTU method as found in Duffie and Beckmann (1991). To simplify the equation, $$\eta = \left(1 - e^{-\frac{\Delta t}{\tau}}\right)$$ is used.

$$T_{n+1} = T_{n,1} \left(1 - \frac{\Delta t}{\tau} \frac{L}{\Delta x} \frac{1}{\eta}\right) + T_{n,1} \left(\frac{\Delta t}{\tau} \frac{L}{\Delta x} \frac{1}{\eta}\right) + \frac{\Delta t}{\tau} \frac{L}{\Delta x}$$

(3.11)

The explicit equation 3.11 is used to determine the temperatures of the rocks in the packed bed.

### 3.3 Thermal losses from the model

Thermal losses through the walls of the test model were accounted for by setting up a thermal resistance diagram as explained by Çengel (2006). The diagram is shown in Figure 8.

For the purposes of the calculations and due to the gradual change in thermal energy with respect to time, steady operating conditions were assumed. The flow inside the bed is assumed to be laminar due to the relatively low Reynolds number. Temperature dependant characteristics of the air were used at the average temperature experienced between the bed and the environment.

Flow along a parallel flat plate was used to calculate the Reynolds number for the length of the bed.
In this equation the velocity \( (V) \), is the absolute velocity in the bed in m/s. The length \( (L) \) of the bed is in meters, and the kinematic viscosity \( (\nu) \) in m\(^2\)/s. The Prandtl number \( (Pr) \) of 0.7282 for air at sea level and at 30 °C is also obtained from Çengel (2006).

\[
R_{SL} = \frac{V L}{\nu}
\]  
(3.12)

The relation above is a modified version of a Nusselt number calculation presented by Çengel (2006) for \( Re_L \) values less than \( 5 \times 10^5 \). It was used to calculate the average heat transfer coefficient between the air in the bed and the polystyrene insulation \( (k_{polystyrene} = 0.040 \, \text{W/m}^2\text{K}) \). It was also used to calculate the average heat transfer coefficient between the plywood \( (k_{plywood} = 0.12 \, \text{W/m}^2\text{K}) \) and the environment.

\[
h = \frac{k}{D} 0.664 Re^0.8 Pr^{0.4}
\]  
(3.13)

The resistance values for the diagram in Figure 8 were calculated using equations 3.14 and 3.15 for the convection and conduction heat transfer coefficients respectively.

\[
R_{conv} = \frac{1}{h_i A}
\]  
(3.14)

In the equation above the total surface area \( (A) \) refers to the internal surface area of the bed. The average heat transfer coefficients \( (h_i) \) of the

\[\text{Figure 8: Thermal resistance diagram of losses from model rock bed}\]
polystyrene (2.54 W/m²K) and the plywood (7.62 W/m²K) were calculated from equation 3.13.

\[ R_{\text{cond}} = \frac{L}{k_A} \]  
(3.15)

The thermal resistances are in series and were summed to calculate the total thermal resistance through the walls of the test bed.

\[ R_{\text{total}} = R_1 + R_2 + R_3 + R_4 \]  
(3.16)

The steady rate of heat transfer was then calculated and incorporated into the analytical calculation of the model test bed.

\[ Q = \frac{\Delta T}{R_{\text{total}}} \]  
(3.17)

The \( \Delta T \) term refers to the temperature difference between the air inside the bed (\( T_{\infty 1} \)) and the air of the surroundings (\( T_{\infty 2} \)). A combined heat transfer coefficient for the losses through the walls of the bed was calculated to be 0.53 W/m²K.

### 3.4 Results of analytical solution of the Model

Microsoft Excel was used to set up the pressure drop and thermal calculations described in sections 3.1, 3.2 and 3.3. These calculations were used to simulate the thermal changes with respect to time. Figure 9 displays the results of a simulated eight hour charging cycle through the test model bed. The sections referred to in this graph are spaced in accordance with the locations of the thermocouples in the test model. See appendix C for details.

The fan and heat supply were then shut down and the thermal behaviour of the bed was monitored. Figure 6 shows the thermal behaviour of the packed bed with no airflow. Sections 1 and 4 show greater thermal losses
than sections 2 and 3 and this can be attributed to the lack of insulation on the inlet and exhaust ducts of the test model.

The input parameters were based on the measured data acquired during the testing of the built model. Section 1 is located nearest the inlet of the packed bed and Section 4 is nearest the exhaust. The figure shows how the packed bed is ‘charged’ in sections over the time period. The systematic charging of the respective sections is referred to by literature as a temperature wave. The temperature curves flatten as the observer moves further from the inlet because the model accounts for thermal losses through the walls of the bed as described in section 3.3. If the losses are ignored the curves have similar gradients throughout the bed.

The results obtained from the discharging cycle were calculated on the same Excel based model. Again, due to thermal losses the gradients of the curves are reduced as we move from sections 1 through 4. The results are based on uniform initial temperature throughout the bed and a constant inlet temperature based on the results of the measured data. The
results of the analytical model during the discharge cycle are shown in Figure 10.

Figure 10: Results from analytical simulation during the discharging cycle
4 Numerical simulation of model test bed

A numerical simulation showing the behaviour of the test model will be investigated in this section. A brief introduction to the methods used by FLUENT in the simulation are included. The model will be set up with similar geometrical proportions to the test model. The core will be specified as a porous medium and will be given the calculated properties. The temperature of the flow through the model will be constant for the duration of the charging cycle. The results will then be compared to the results obtained from the analytical analysis presented in chapter 3.

4.1 Description of the numerical simulation

A single phase porous media model was selected for the simulation of flow through the packed bed. A cell zone was defined as the porous section so that heat transfer through the model could be determined. The heat transfer represented was based on the assumption that thermal equilibrium between the porous medium and the flow exists. In order to prevent this assumption a multiphase model would need to be specified and a much greater understanding of the workings of computational fluid dynamics would be required.

4.2 Limitations and assumptions of the numerical simulation

A porous media model calculates the added flow resistance created by the packed bed by adding a momentum sink to the empirical momentum equations.

The volume occupied by the rocks in the packed bed is not represented in the numerical simulation. To compensate for this a superficial velocity is calculated for the flow through the defined porous region. The superficial velocity is a relation between the specified true velocity and the porosity of
the medium. A constant volumetric flow rate is assumed through the porous region in order to maintain continuity in the velocity vectors.

### 4.3 Momentum equations in the numerical simulation

The added momentum source is shown in equation 4.1. The viscous loss term is standard to the fluid flow equation and was initially presented by Darcy. The internal loss term is the second term on the right hand side of equation 4.1 and the sum of the two make up the source term \((S)\).

\[
S_z = -\left( \sum_{j=1}^{n} D_{ij} \frac{\partial u_j}{\partial z} + \sum_{j=1}^{n} C_{ij} \frac{1}{2} \|u\| u_j u_j \right)
\]

(4.1)

The source term is valid for the \(x\), \(y\) and \(z\) directions. The magnitude of the velocity is represented by \(\|u\|\). Prescribed matrices are represented by \(D\) and \(C\) in the calculation of the source term. The loss of momentum in the porous cell is created by calculating a pressure drop, proportional to the velocity of the fluid, in the cell.

Equation 4.1 can be simplified to the relation shown in equation 4.2 if the porous medium is homogeneous, as is the case for the packed bed.

\[
S_z = -\left( \frac{K}{\alpha} u_z + C_2 \frac{1}{2} \|u\| u_z \right)
\]

(4.2)

\(C_2\) refers to the internal resistance factor of the packed bed. The permeability of the bed \((\alpha)\) is also required. The simplification of equation 4.1 can also be obtained by specifying \(D\) and \(C\) as diagonal matrices. These matrices should have \((1/\alpha)\) and \((C_2)\) as the values of the diagonals in \(D\) and \(C\) respectively. Due to the low velocity present in the packed bed, laminar flow is assumed. Thus the pressure drop, which is proportional to the constant \(C_2\) and the velocity, is considered to be zero. FLUENT then calculates the pressure drop in the \(x\), \(y\) and \(z\) directions by using the equation 4.3, which is directly formulated from Darcy’s Law.
\[ \Delta \bar{P}_x = \sum_{j=1}^{3} \frac{\mu}{\alpha_{x,j}} v_j \Delta \bar{h}_{xx} \]  

(4.3)

The equation above represents the change in pressure over the cell in the \( x \) direction. The same form of equation is valid for directions \( y \) and \( z \), with appropriate changes made to the subscripts. The thickness of the medium \( \Delta n_l \) refers to the actual thickness of the model in the respective directions.

### 4.4 Energy equation in the numerical simulation

The energy equation in the porous media model is altered from the standard energy transport equation. An effective conductivity is used in the porous zone. The thermal inertia of the solid region is included in the transient term of the standard energy transport equation. The resultant energy transport equation is shown in equation 4.4.

\[
\frac{\partial}{\partial t} \left( \rho_f E_f + (1-\gamma) \rho_s E_s \right) + \nabla \cdot \left( \overline{\rho_f} E_f \nabla \bar{T} + \rho_s \overline{E_s} \nabla \bar{T} \right) = \nabla \cdot \left( - \sum_i \overline{k_i} \nabla \bar{T}_i \right) + \overline{S_B} 
\]

(4.4)

Where \( \langle \overline{E_f} \rangle \) refers to the total fluid energy. The total solid medium energy is represented by \( \langle E_s \rangle \), and \( \gamma \) is the porosity of the medium. The fluid enthalpy source term is \( \langle S_B \rangle \), and the effective thermal conductivity of the porous medium is \( \langle k_{eff} \rangle \). The calculation of the effective thermal conductivity is shown in equation (4.5).

\[
k_{eff} = \gamma k_f + (1-\gamma) k_s
\]

(4.5)

Where \( \langle k_f \rangle \) is the thermal conductivity of the fluid and \( \langle k_s \rangle \) is the thermal conductivity of the solid mass in the porous zone.
4.5 Methods used in the numerical simulation

This section provides details on the settings and inputs used to run the numerical simulation of the model. The model was set up as a three dimensional, transient, laminar, pressure based model.

The rectangular shape of the model was created and dimensions were added to match those of the experimental test bed. The model was meshed with minimum node sizes of 50 mm. A “successive ratio” of 1.2 was added to the nodes near the ends of the model where the surfaces would be monitored during the simulation.

Double precision was selected for the solver. The time set to transient with second order implicit solver selected. The viscous model was set to laminar and the energy equation enabled.

Named sections were added to the mesh as shown in Figure 11 and the boundary conditions were specified. The named “Inlet” was specified as a “velocity-inlet” with the inlet air’s physical velocity set to 0.1 m/s and the temperature of the inlet air set to 323 K.
The “Wall” was specified as a “wall” and referred to the four outer insulated sections of the model. A thermal conductivity value of 0.53 W/m²K was specified and the wall thickness was set to 0.06 m. The “Outlet” boundary condition was set to “pressure-outlet” with gauge pressures set to zero.

The fluid was specified as air with a density of 1.225 kg/m³ and a viscosity of $1.789 \times 10^{-5}$ kg/ms. The specific heat capacity was set to 1006.43 J/kgK and the thermal conductivity to 0.0242 W/mK. The molecular weight of the fluid was 28.966 kg/kmol. A solid medium was defined and named “Berea” with a specified density of 2150 kg/m³. The specific heat capacity of the solid was set to 850 J/kgK and thermal conductivity specified as 2.9 W/mK.

A porous zone was specified and the relative velocity resistance formulation was selected. A porosity of 0.4 was specified for the interior
solid of the model. The initial temperature of the interior solid was set to 288 K.

The solver type was selected as “Green-Gauss node based”. The energy and flow equations were selected for the solver and time steps were set to 60 seconds. The number of time steps was set to 480 in order to simulate the 8h charging cycle of the physical test model. The pressure and momentum relaxation factors were set to 0.3 and 0.7 respectively. The density, body forces and density relaxation factors were left at the default value of one. The pressure velocity coupling parameter was set to “SIMPLEC”. The pressure discretization scheme was set to “PRESTO” and both the momentum and energy discretization schemes were selected as “Second Order Upwind”.

A surface area monitor was set up on both the inlet and outlet boundaries. The “Area weighted average” of the “Static temperature” of both of these surfaces was captured at each time step. The results of the outlet boundary temperatures are plotted in Figure 12. The analytically determined results from the outlet of the model packed bed described in chapter 2 are shown as a comparison to the numerical solution.
4.6 Discussion of the results obtained from the numerical solution

The initial variations between the analytical and numerical results displayed in Figure 12 could be attributed to the fact that the porous media model assumes that the temperature of the fluid and the solid are equal at any point along the porous region. The increased temperature of the analytical model is due to the difference in the air and rock temperatures as the air does not transfer all of the available thermal energy to the rocks immediately. This could be accounted for in the numerical model by setting up a multi-phase numerical model. This model could be set up to take the size of the rocks into account, and also does not assume that the solid and fluid temperatures are equal throughout the porous region.

The decrease in the temperature difference between the analytical and numerical models later in the curves is more difficult to explain. Further research into the way the numerical value is calculated is recommended to explain this variation.
Due to the variations seen between the analytical and numerical models shown in Figure 12, the rock store at the US Sustainability Institute will not be numerically modelled for the purposes of this report. Further investigation into multi-phase numerical simulation is recommended to increase the accuracy of the numerical results.
5 Comparative results

This chapter compares the results obtained from the measurements taken at Lynedoch as well as the test model, to the predicted results obtained from the analytical and numerical solutions. The accuracy of the analytical and numerical models are discussed and reasons for irregularities are investigated.

5.1 Comparative results of the model

This section shows comparisons between the analytical calculations and the measured results in graphical format. The results of the measured data, for Figure 13 through to Figure 16, are represented by dotted lines and the analytical curves are shown by the solid lines. Figure 13 shows the comparison for first section of the model. This section is located nearest the inlet. Figure 16 shows the comparison for the fourth section which is located nearest the exhaust of the test bed.

Figure 13: Comparative graph of the Model - Section 1
The analytical results in Figure 13, predict an increase in temperature sooner than the measured results. This could be due to losses not accounted for in the inlet duct section of the measured model which was not insulated in the same manner as the walls of the test bed.

![Graph](image)

**Figure 14: Comparative graph of the Model - Section 2**

The comparative results, shown in Figure 14 and Figure 15, show that the areas of the test model which were insulated to minimise thermal losses. The results show favourable comparisons between the measured data and the analytical calculations.
Figure 16 shows the comparative results for the final section of the test model. The analytical results predict a slower rise in temperature than the measured data. This could be due to the porosity of the analytical model being slightly lower than the porosity in the test bed.

The temperature results from the calculated data show a total increase in thermal energy over a longer period of time than the results of the measured data. High levels of humidity were experienced during testing.
but Chandra and Willits (1981) commented that humidity does not have a substantial effect on the packed bed. The volumetric heat transfer coefficient sourced in literature, of 2.9 W/mK, may be slightly lower than the actual heat transfer coefficient of rocks used in the test model packed bed.
5.2 Lynedoch

The primary objective of this report was to set up an accurate analytical model of the rock store located at the US Sustainability Institute at Lynedoch. This section of the report provides a description of the details of this rock store as well as information regarding the collection of data from the facility.

The dimensions and specifications of the rock thermal storage system at the Sustainability Institute, along with Figure 17 showing the construction phase, were obtained from Mr Alastair Rendall of ARG Design.

Figure 17: Construction of the rock store at the Sustainability Institute

The rock store measures 40 m in length, is 3 m wide and 0.5 m deep. The store is filled with rocks which were obtained from nearby the Sustainability Institute. The rocks are sandstone (Berea), which are predominantly made up of quarz. The rocks have an average cross-
sectional diameter of 150 mm. Air is pumped through the rock store by two fans which are connected to Ø300 mm ducts. The exhaust air from the rock store is channelled into five classrooms above the rock store. The air enters each classroom via four 400 x 230 mm exhaust grills. The combined volume of the classrooms is 600 m³. Each classroom has a large, north facing window which is shielded from direct sunlight. The air supply to the fans is either drawn from the roof above the classrooms or from directly outside the fan room. Air is drawn from the roof, during the day, when the rock store is used for heating purposes. In the cooling cycle, cold air is drawn from directly outside the fan room at night. The system can switch between modes by an automated, programmable shutter system shown in Figure 18.

![Figure 18: Supply fans and automated shutter mechanisms](image)

This automatic system is currently not in use and the system is operated manually.

Measurements from the Sustainability Institute were taken at the exhaust grills in the classrooms over a period of one week during simulated test of
both the ‘summer’ and ‘winter’ cycles. The temperature measurements were made at half hour intervals with a digital hand held thermometer. These were compared to the ambient temperatures at the time. The inlet and outlet temperatures are shown in Figure 21 and Figure 22 of this report.

5.3 Analytical data for Lynedoch

The analytical model developed in Microsoft Excel was set up to match the geometry of the rock store at the Sustainability Institute at Lynedoch. The analytical model was then run for both the charging and recovery phases with constant input temperatures. The theoretical results of these simulations are shown in Figure 19 and Figure 20.

![Figure 19: Analytical results for simulation of Lynedoch charging cycle](image)

Arbitrary constant inputs, shown in the figures, were selected so that the analytical behaviour of the bed could be observed more carefully.
The results of the charging cycle are shown in Figure 19 for an initial bed temperature of 20 °C with an ambient environment of 30 °C. The temperature of the first section rises the fastest, and the temperature of the final section rises slowest. This is referred to as a temperature wave in a packed bed. The slope of the temperature wave is largely dependent on the hydraulic diameter of the rocks used in the packed bed. Smaller rocks would cause the temperature wave to become steeper.

![Figure 20: Analytical results for simulation of Lynedoch-recovery cycle](image)

The theoretical recovery cycle has a constant input and environmental temperature of 15 °C. It is assumed that the bed has a uniform initial temperature of 27 °C as shown in Figure 20. The temperature wave flattens further along the test bed. The flattening of the temperature wave can be attributed to the relatively large rocks used in the packed bed.

The curves for both the charging and discharging cycles are similar, with the most noticeable difference being the rate at which the store discharges. The store discharges faster than it charges due to thermal
losses from the bed accelerating the discharging cycle and decelerating the charging cycle.

5.4 Comparative results for Lynedoch

Temperature readings were taken from the centre exhaust duct at Lynedoch at half hour intervals during the mornings of the 21\textsuperscript{st} and 23\textsuperscript{rd} of September 2010. The measured data is then compared to the outside ambient temperatures and the analytical predicted exhaust temperatures of the rock store. These results are plotted in Figure 21 and Figure 22. The fan pumping air over the rock store was running between 07:00 and 17:00 for the period during which the measurements were taken.

![Graph showing comparative results](image)

**Figure 21: Comparative results of Lynedoch for 21-09-2010**

In Figure 21 the analytically predicted results resemble the measured data quite accurately. The reason for the increase in the exhaust temperature of the rock store at 07:00 is due to incomplete charging from the previous
day. The inlet side of the packed bed had more stored thermal energy than the exhaust side. After the fan was turned on, the energy stored in the bed causes the exhaust temperature to rise as the thermal wave moves through the bed. As the ambient temperature increases, the thermal energy is absorbed by the packed bed. The fan is shut off at 17:00 after the rock store has started charging with the warm afternoon air. This energy is stored by the bed until the fan is turned on again the next morning. These results are an indication of how the rock store can be used in winter to store energy in the afternoon when the ambient temperature is at its highest and discharge this energy the next morning to increase the room temperature.

![Figure 22: Comparative results of Lynedoch for 23-09-2010](image)

Figure 22 show similar results to those obtained in Figure 21. The analytical results were determined by simulating air flow over the rock store between 07:00 and 17:00 for a five day period. The measured results were then inserted at times they were taken and the measured
data plotted against the predicted behaviour. The results show that the analytical predictions are accurate enough to allow the analytical model to be used when designing a rock store.
6 Recommendations

This section makes use of the analytical method described in Chapter 2. The deviations of the measured results from the theoretical were small enough that the predictions could be made. Predictions of possible performance levels that could be achieved for the hottest and coldest weeks experienced in Lynedoch in 2010 are graphically displayed in Figure 23 and Figure 24 respectively. The recommendations are broken down into five phases of increasing complexity.

6.1 Phase 1: Basic recommendations

Training of the staff responsible for the operation of the rock store at Lynedoch is recommended. The operators should be informed on how the system functions including required charging times. The duration of time that benefits can be expected from the system for particular conditions should be incorporated into the training too. The input selection for particular ambient conditions should also be carefully explained so that the operator can fully understand the implications of the selected input with regard to the desired outputs.

6.2 Phase 2: Gaining maximum benefit from the rock store during the summer months

Lynedoch is prone to experiencing extremely hot summers. Rainfall during the summer is very scarce and this contributes to the relatively large variation between daily maximum and minimum temperatures. These conditions allow the packed bed to be cooled at night and to release cooler than ambient air during the day. Figure 23 shows the comparison between the ambient and packed bed exhaust temperatures for one of the hottest weeks experienced in 2010 thus far. This prediction assumes that the fans to the rock store are kept running constantly.
Under these conditions it can be noted that the rock store can continually supply cooler air to the rooms between 07:00 and 17:00. The outlet temperature is dependent on the ambient temperature of the previous night. As the minimum night temperature is usually experienced shortly before sunrise, the charging period between midnight and sunrise is the most crucial if the rock store is to provide cooled air the following day.

This recommendation requires only that the fans run constantly. At a mass flow rate of 2.4 kg/s through the fans, the air flowing over the rock store should be sufficient to completely replace the air in the rooms in a fraction under five minutes.

During the week shown in Figure 23, the total amount of thermal energy removed from the rooms between 07:30 and 17:00 amounts to 4.86 GJ. The two fans are each rated at 0.55 kW and thus require 0.475 GJ of electrical energy to run constantly for a five day period. As a measure of comparison, for the period of the week shown, the ratio of energy output over energy input is 10.22. This is calculated as one would calculate the coefficient of performance of a refrigeration cycle, or an air-conditioner. It is, however, unfair to compare the rock store to an air-conditioner as it relies completely on ambient conditions and although it can provide active cooling it cannot control humidity.
Figure 23: Predicted performance for operation during summer

6.3 Phase 3: Gaining maximum benefit from the rock store during winter months

Contrary to the advantageous summer weather conditions, Lynedoch experiences high rainfall during the winter. The increase in humidity has a direct effect on the variation between maximum and minimum daily temperatures. These small temperature fluctuations impact negatively on the benefits that can be expected from the rock store. The results are not entirely negative, and even limited management of the fans can contribute greatly towards the thermal benefits that can be expected.

Figure 24 shows a comparison between the ambient temperature and the exhaust temperature of the rock store. The predicted exhaust temperature assumes that the fans are only running between 07:00 and 17:00 daily.

This allows the rock store to charge during the day, particularly in the afternoon when the ambient temperature is at its highest. At 17:00 the
fans are shut down and the thermal energy is stored in the rock store. At 07:00 when the fans are started up, the stored energy is pumped into the rooms. Figure 24 shows that the exhaust temperatures increase on the Wednesday and Friday of the week shown. This is due to incomplete charging of the rock store the previous day. The inlet side of the rock store has more thermal energy than the exhaust side when the fans are shut down. The peaks show the transfer of thermal energy through the bed.

![Figure 24: Predicted performance for operation during winter](image)

The ambient temperature shown in the comparison is taken from the coldest week experienced during 2010. The comparison shows that the greatest benefits can be expected if an extremely cold day follows a moderate day. Although the overall heat supplied is not excessive, the rock store does contribute positively early in the mornings when the ambient temperature is at its coldest.
6.4 **Phase 4: Incorporating automatic timers to start up and shut down of the fans**

The next phase of the recommendations requires that the timer controlled start-up and shut down system installed at the facility be repaired.

The system can then be optimised by programming the start-up and shut down times of the fans. This will reduce the energy consumption of the system while keeping the benefits gained from the system to a maximum.

The installed system also has the option to select where the inlet air is drawn from. During the summer months when cooling is required the system will get its intake air directly from vents in the walls of the fan room. During winter operation the system can select to draw the inlet air from a thermal catchment system which is built into the roof cavity of the building.

Overcast and rainy conditions were experienced when measurements were taken of the performance of the system when the inlet air was drawn from the roof. As such the results did not show any significant advantages of collecting the inlet air from the roof. Further investigation into the performance of the solar thermal collector in the roof would be able to establish the benefits that can be expected.

6.5 **Phase 5: Fully automatic operation of the fans**

The fifth phase of the recommendations requires constant monitoring of the temperatures in the rock store as well as the ambient temperature.

Further investigation, based on the principals explained in this report, could be used to set up active control of the fans. This system could then be optimised to gain the maximum benefit from the rock store while consuming the least electrical energy. The system could be programmed to make active input selections and operate the fan when required by monitoring the temperature differences between the rock store and the ambient conditions.
Included in this recommendation is the installation of ducts which would allow the exhaust air from the rock store to bypass the classrooms. This would allow the rock store to be charged without the exhaust air affecting the classroom. In the winter months this will allow air flow from the rock store bypass the classroom as soon as the exhaust temperature drops below the temperature in the classroom. This would be the ideal method of operation for the thermal rock storage facility at Lynedoch.
7 Summary

The thermal and fluid dynamics analysis of the US Sustainability Institute’s School of Public Management and Planning under floor thermal rock storage facility was the primary objective of this project.

In order to achieve this objective careful study of the available literature relating to thermal storage in packed beds was done. The scope of the project was then defined.

This scope included the design, construction and testing of a scaled model of the rock storage facility at Lynedoch.

An analytical model was set up to simulate the results of the test model. These comparisons proved the accuracy of the analytical model.

Physical measurements were then taken at the rock store located at Lynedoch. The geometry and fluid flow rate of the analytical model was altered to match that of the packed bed at Lynedoch. The similarities between the analytically predicted results and the measured data satisfied the primary objective of the project.

A numerical model was then created using a single phase porous medium in FLUENT. The results of the computational fluid dynamics did not match the analytically predicted results for the test model. It was concluded that a multi-phase porous media model would provide better results for the numerical simulation, but this fell beyond the scope of the project.

A series of recommendations for the operation of the rock store at Lynedoch was presented along with the results that could be expected should the rock store be operated as recommended.
8 Conclusions

The test model that was designed and constructed provided acceptable results. These could be further improved if the inlet and exhaust ducts were insulated in the same manner as the bed was. The testing of the model was conducted during winter, which resulted in artificially large thermal losses. The model could be tested under more suitable ambient conditions, for example in summer. This would allow ambient air to be used for the inlet, resulting in reduced thermal losses through the walls of the bed, and allowing greater accuracy in the timing of the charging and recovery phases of the rock store.

The analytical model which was set up for the purposes of this project yielded successful results compared to the measured data. The model is capable of working with a variety of input data and varying time frames. The analytical model could be used successfully for both the model of the rock store and the rock store at Lynedoch.

The simplified approach used for the numerical simulation did not show accurate results. A multiphase porous media model is recommended as it would increase the accuracy of the simulation which would increase the value of the results.

The analytical model allows accurate results to be predicted which can be used to optimise the operation of the rock storage facility at Lynedoch.

The rock storage facility at Lynedoch is currently not being used to its fullest potential. Greater rewards can be gained from the system if the operation of the system is managed more carefully.
References


Appendix A: Project cost section

A breakdown of the cost of the project is included in Table 1. Some of these costs include estimations as the costs of certain aspects of the project were difficult to determine. Some of these aspects include the value of guidance received from experts in the field and software licences.

It is also difficult to put a financial value on the contribution this project has made to the client. The project can contribute to the comfort level experienced in the classrooms located above the thermal rock storage facility.

The project could also be used should further research into thermal rock storage systems as part of ventilation systems be desired. The project did serve as proof that the explicit method presented by Allen (2010) to solve thermal relations in rock storage systems is applicable even though relatively large rocks were used.
Table 1: Details of the costs related to the project

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<th>Facility Use</th>
<th>Capital Costs</th>
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## Appendix B: Project Gantt Charts

### Figure 26: Predicted planning for the project

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<tr>
<th>Task Name</th>
<th>Duration</th>
<th>Start</th>
<th>Finish</th>
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<td>1 Literature Study/Research</td>
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<td>Tue 2/21/10</td>
<td>Mon 3/29/10</td>
</tr>
<tr>
<td>2 Identify Project Requirements</td>
<td>10 days</td>
<td>Tue 3/2/10</td>
<td>Mon 3/23/10</td>
</tr>
<tr>
<td>3 Generate Model Concepts</td>
<td>10 days</td>
<td>Tue 4/6/10</td>
<td>Mon 4/20/10</td>
</tr>
<tr>
<td>4 Acquire System Data and Measurements from Lynedoch</td>
<td>20 days</td>
<td>Tue 4/10/10</td>
<td>Mon 5/4/10</td>
</tr>
<tr>
<td>5 Refine Model Concepts</td>
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<td>Tue 6/2/10</td>
<td>Wed 6/16/10</td>
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<tr>
<td>6 Build Model</td>
<td>30 days</td>
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<td>Wed 7/7/10</td>
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<td>7 Source Testing Equipment</td>
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<td>Thu 7/9/10</td>
<td>Wed 7/29/10</td>
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<tr>
<td>8 Mathematical Analysis of Model</td>
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<td>Thu 7/29/10</td>
<td>Tue 8/24/10</td>
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<td>9 Test Model</td>
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<td>Mon 9/18/10</td>
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</tr>
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<td>16 Write Final Report</td>
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</table>

### Figure 27: Actual task duration for project

![Gantt Chart Diagram](chart.png)

Figure 26: Actual task duration for project
Appendix C: Layout of thermocouples in model

Figure 27: Instrumentation positioning for test model